TITLE: DEVELOPMENT OF A COMBINED DIESEL EXHAUST COOLER/SCRUBBER FOR USE IN UNDERGROUND OPERATIONS

FINAL REPORT

BY

Hovey & Associates (1979) Ltd.
2378 Holly Lane
Ottawa, Ontario
K1V 7P1

Hovey File 2002

Work on this project was conducted under the auspices of the Canada Centre for Mineral and Energy Technology, Energy, Mines and Resources, Canada.

DSS FILE NUMBER: 23SQ.23440-2-9180
DSS CONTRACT SERIAL NUMBER: 1SQ.82-00266
CANMET PROJECT NUMBER: 23440-2-9180
CANMET SCIENTIFIC AUTHORITY: Mr. E. D. Dainty

OCT 26 1984
MICROMEDIA

41 AP

TABLE OF CONTENTS

1.0 AIM

2.0 SUMMARY

3.0 TECHNICAL DISCUSSION

- 3.1 Fabrication
- 3.2 Assembly
- 3.3 Modifications
- 3.4 Change in Program Scope

APPENDIX "A" - ASSEMBLY PROCEDURE

APPENDIX "B" - FLOW METER SPECIFICATIONS

FINAL REPORT 2002 RE0010

1.0 AIM

1.1 This project was to design and build a prototype cooler/scrubber to determine the benefit of reducing water consumption by precooling exhaust gases when using a water bath type scrubber.
The design calculations were described in the Phase I report. A description of the fabrication and assembly are included in this report.

2.0 SUMMARY

2.1 The fabrication of the prototype cooler scrubber is described including the modifications to provide access for test equipment and viewing of the 'scrubbing' action.
The change in scope of the program necessitated by dynamometer scheduling is presented along with a list of test equipment purchased on the program.

3.0 TECHNICAL DISCUSSION

3.1 Fabrication

The majority of parts were bought in from the steel supplier plasma arc cut to size to decrease the shop labour required. As the tolerances were not held, the plates were returned for rework which delayed the program by three weeks.
Fabrication of the components was by normal shop machining and welding practices. All parts were inspected and test fit to the next assembly before welding.

.../2

3.2 Assembly

Due to the complicated nature of the baffles and stiffeners, a step by step procedure was developed to assemble the cooler/scrubber. This is presented as Appendix A.

3.3 Modifications

Modifications to the cooler/scrubber consisted of the addition of windows in the side panels and fittings welded to the panels to mount test equipment.

The window near the bottom provides a view of the outlet of the nozzle where the gases are discharged into the water. The nozzle on the opposite side provides a view of the baffles used to separate the entrained water from the exhaust gases.

Parts were fitted into the side panels, and manifolds at certain locations as shown in Figure 1. The parameters to be measured are listed below.

Part No.	Parameter	
1	Temp, Press,	Cooling water inlet
2	Temp, Press,	Cooling water outlet
3	Temp, Press,	Exhaust gas, inlet to cooler
4	Temp, Press,	Exhaust gas, cooler outlet
5	Temp	Scrubber water
6	Temp	Scrubber water
7	Press,	Gas flow
8	Temp, Press,	Gas, scrubber outlet

At the conclusion of the test program, the windows and ports can be welded up, prior to explosion proof testing.

.../3

3.4 Change in Program Scope

The program scope was changed as the dynamometer could not be schedule to be made available for this test program.

The activities listed as part of our requirements which were deleted as a result of this were the observations of the testing, and modifications during testing. In lieu of these, some test material and equipment was purchased by EMR on the program which will be required for the dynamometer testing. In addition, the contract price has been reduced to reflect the deletion of these activities.

The additional test material and equipment is listed below:

(1) Water flow meter for measuring coolant flow rates. Technical data attached as Appendix B.

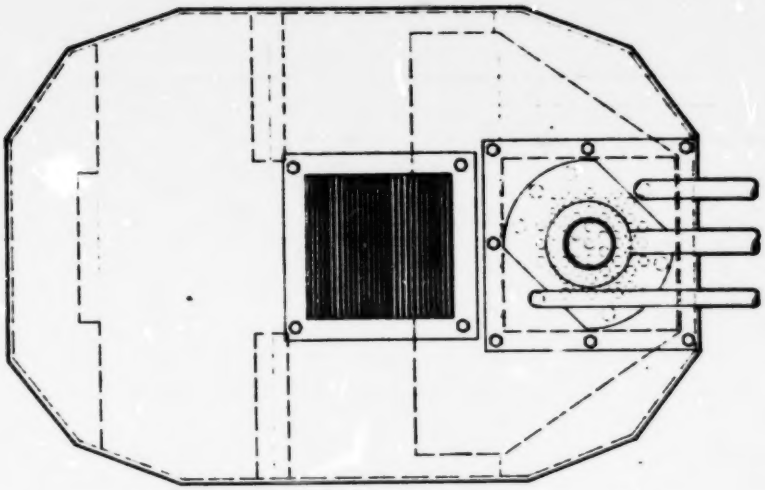
(2) Plumbing materials for routing of coolant water:
- 30 ft of 1-1/4 inch copper tubing
- 20 ft of 1-1/4 inch flexible braided water hose
- 10 ft of 1 inch flexible braided water hose
- reducers
- hose clamps
- 1 gate valve 1-1/2 inch
- 1 globe valve 1-1/2 inch
- 4, 1/4 NPT tees for temperature and pressure probes
- 200 ft of 1/4 inch tygon flexible tubing
- Tees, 1-1/2 NPT

APPENDIX "A" to
Final Report
File 2002

ASSEMBLY PROCEDURE

Assembly Procedure - Cooler/Scrubber
Item Nos. refer to items on drawing 3053-0001

1. Weld body panel halves together.
2. Weld baffles 6 & 11 to top plate 4.
3. Weld top plates 3 & 4 together.
4. Weld baffles 10 to top plate, 4.
5. Weld stiffeners 15 & 16 to body panels.
6. Assemble body onto top plate, upside down.
7. Weld main baffle 7 into scrubber.
8. Plug weld 10 & 11 through holes in 7.
9. Stitch weld stiffener 17 & 18 to walls - leave space where stiffeners 13 must be positioned.
10. Weld baffle 9 to walls.
11. Weld stiffeners 13 to 17, but do not install stiffeners onto 18.
12. Install baffle 12 nearest nozzle. Do not install other item 12.
13. Install 3 stiffeners 14 evenly spaced across scrubber.
14. Insert splitter 8 into nozzle on 6 and weld to inside of body and item 14.
15. Weld 3 of stiffeners 13 onto vertical stiffeners 18.
16. Install second baffle 12 onto walls.
17. Install drain couplings 23 into walls.
18. Weld bottom plate 2 onto walls.
19. Install stiffener channels 20, 21, 22.



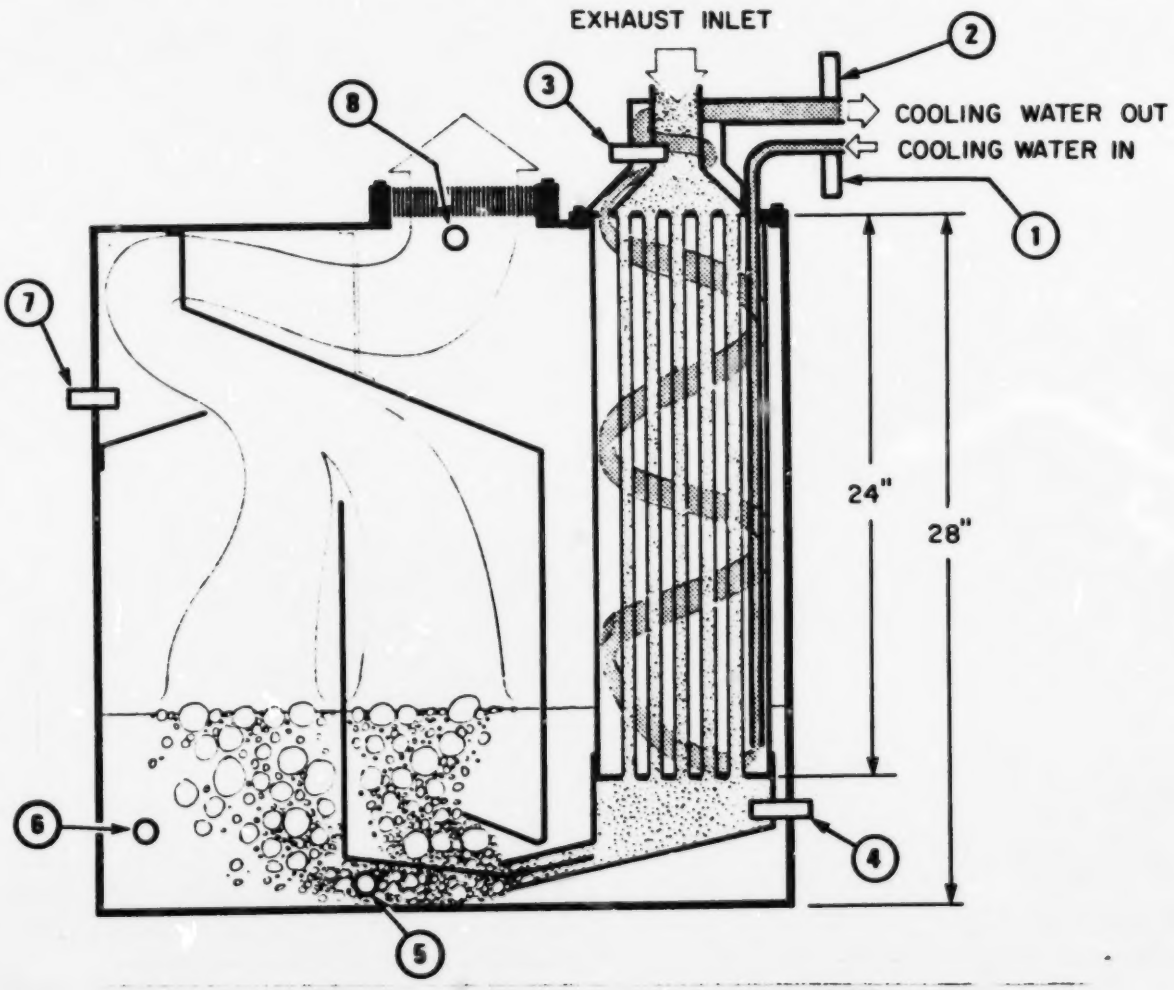


FIGURE 1 COMBINED COOLER SCRUBBER

HEDLAND

flow meters for liquids air or gases

HEDLAND Liquid Flow Meters

- ☐ **NO ELECTRICAL CONNECTIONS**
- ☐ **GOOD VISCOSITY STABILITY**
- ☐ **ACCURACY WITHIN ± 5%**
- ☐ **ONE YEAR LIMITED WARRANTY**
- ☐ **DIRECT READING**
- ☐ **LOW COST**
- ☐ **RUGGED**

HEDLAND LIQUID FLOW METERS are direct reading units requiring no electrical connections or readout devices. Available in 6 sizes, the meters can be used at pressures up to 3000 PSI (800 PSI for the 3" meter and 250 PSI for the EZ-View meter) and are relatively insensitive to shock, vibration, viscosity and mounting position. Each size is available with several different flow rates ranging from a low of .05 to .5 GPM for the 1/4" meter to a high of 20 to 300 GPM for the 3" meter.

Standard HEDLAND Flow Meters can be furnished to measure either the rate of flow for oil (specific gravity of .84) or water (specific gravity 1.0). Flow rate is a function of specific gravity, therefore either a conversion curve or special scales could be required for measuring the rate of flow of fluids with specific gravity other than .84 or 1.0. For compatibility with other fluids, it should be noted that the standard Flow Meters are made of aluminum (5 sizes) or optional brass (3 sizes).

A 1" EZ-View Flow Meter is also available for low flow, low pressure applications (see page 5 for details).

HEDLAND PNEUMATIC FLOW METERS are also direct reading units requiring no electrical connections or readout devices.

Available in 4 sizes, the meters can be used at pressures up to 600 PSI. Several different flow ranges are available from a low of 4 to 24 SCFM for the 1/4" size to a high of 40 to 280 SCFM for the 1-1/4" size.

Scales are calibrated to a temperature of 70° F and 100 PSI. A correction factor chart for other temperatures and pressures is located on each meter for convenience.

Standard meters are made of aluminum. Optional meters in brass (3 sizes) are also available.

SPECIAL SCALES Please consult factory if your flow measuring requirements call for special scales for different specific gravities, special fluids or gases or different increments of measure.

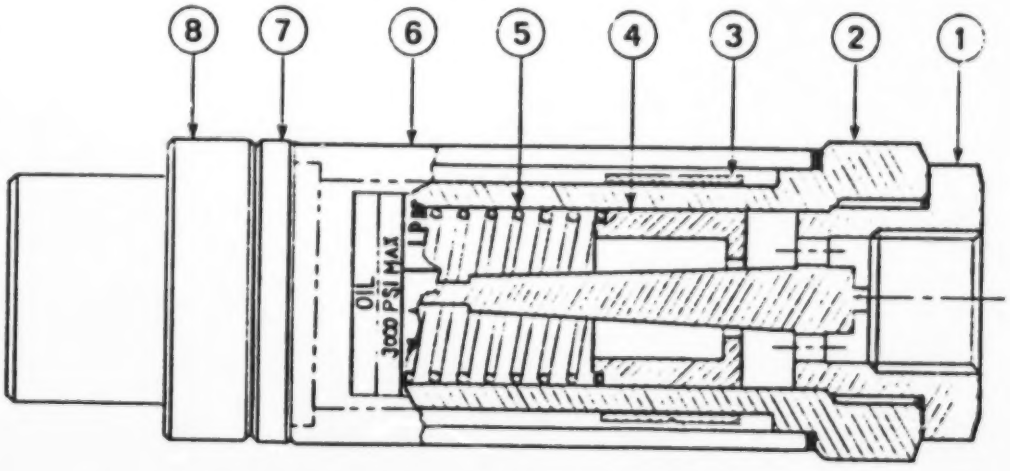
OPERATING PRINCIPLE

Requiring no electrical connections or readout devices, the Hedland Flow Meters use an annular orifice formed by a piston and a tapered cone. The piston is held in place at the base of the cone (no flow position) by a calibrated spring. Flow causes a difference in pressure that moves the piston against the spring. Piston displacement is directly proportional to changes in flow rate.

A magnet is fitted on the piston that moves the steel indicator ring, located outside of the non-ferrous metal flow meter housing. The black line on the indicator ring is read against the scale which is located on the transparent dust cover.

The meters utilize a sharp edged orifice. Therefore, variances in viscosity has a negligible effect on accuracy.

Scales are based on specific gravities of .84, for oil meters, and 1.0 for water meters. For fluids with other specific gravities, correction factors can be supplied or special scales can be installed, at a modest additional charge.



1 CAP & METERING PIN ASSEMBLY
2 HIGH PRESSURE BODY
3 FLOW INDICATOR
4 PISTON ASSEMBLY
5 SPRING
6 DUST GUARD & SCALE ASSEMBLY
7 DUST GLAND
8 RETAINER GLAND

Copyright 1982
HEDLAND PRODUCTS
Div. Racine Federated
2200 South St.
Racine, Wisconsin 53404

HEDLAND In-Line Flow Meters

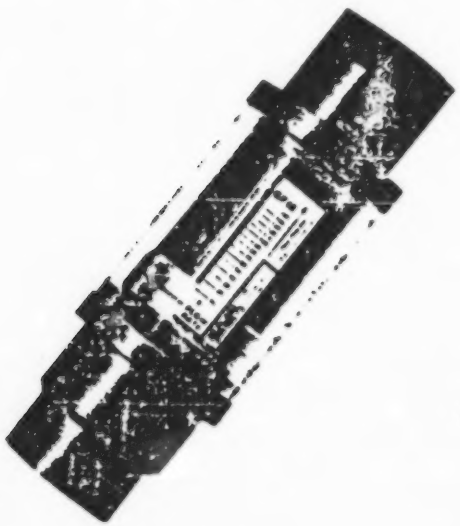
3/4" SERIES

Pressure Rating: 3000 PSI Max. 3:1 Safety Factor
Temperature: 240°F Max. Standard
400°F Optional (consult factory)
600°F Optional (consult factory)
Pressure Drop: See chart page 6
Ordering Information:

Meter No.	Flow Ranges*	Material	Type Sp. Gr.**	Thread
700-002	0.1 - 2 gpm	Aluminum	Oil/.84	#12 SAE
700-005	0.5 - 5 gpm			
700-010	1.0 - 10 gpm			
700-015	1.0 - 15 gpm			
700-020	1.0 - 20 gpm			
700-030	2.0 - 30 gpm			
701-002	0.1 - 2 gpm	Aluminum	Oil/.84	NPT
701-005	0.5 - 5 gpm			
701-010	1.0 - 10 gpm			
701-015	1.0 - 15 gpm			
701-020	1.0 - 20 gpm			
701-030	2.0 - 30 gpm			
703-002	0.2 - 2 gpm	Aluminum	Water/1.0	NPT
703-005	0.5 - 5 gpm			
703-009	1.0 - 9 gpm			
703-012	1.0 - 12 gpm			
703-018	1.0 - 18 gpm			
703-025	2.0 - 25 gpm			
705-002	0.2 - 2 gpm	Brass	Water/1.0	NPT
705-005	0.5 - 5 gpm			
705-009	1.0 - 9 gpm			
705-012	1.0 - 12 gpm			
705-018	1.0 - 18 gpm			
705-025	2.0 - 25 gpm			

Material: Standard — Aluminum
Optional — Brass
Size: 3/4" NPT or #12SAE Ports
2 1/4" O.D. 7 5/32" Length

1 1/4" SERIES



Pressure Rating: 3000 PSI Max. 3:1 Safety Factor
Temperature: 240°F Max. Standard
400°F Optional (consult factory)
600°F Optional (consult factory)
Pressure Drop: See chart page 6
Ordering Information:

Meter No.	Flow Ranges*	Material	Type Sp. Gr.**	Thread
801-030	3 - 30 gpm	Aluminum	Oil/.84	NPT
801-050	5 - 50 gpm			
801-075	10 - 75 gpm			
801-100	10 - 100 gpm			
801-150	5 - 150 gpm			
803-027	3 - 27 gpm	Aluminum	Water/1.0	NPT
803-050	5 - 50 gpm			
803-070	5 - 70 gpm			
803-090	10 - 90 gpm			
803-135	10 - 135 gpm			

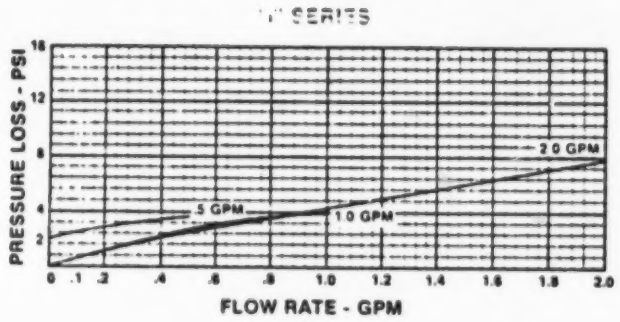
Material: Standard — Aluminum
Size: 1 1/4" NPT Ports 4" O.D. 12 3/16" Length

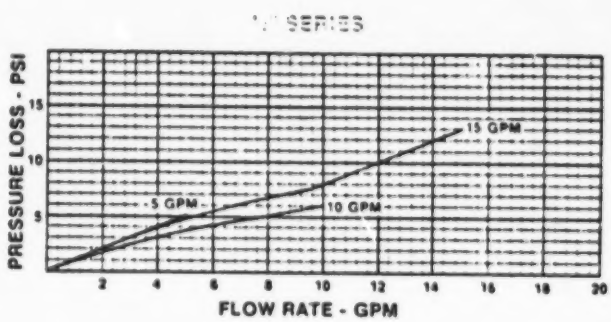
Scales can be calibrated in other units of measurement to fit applications.

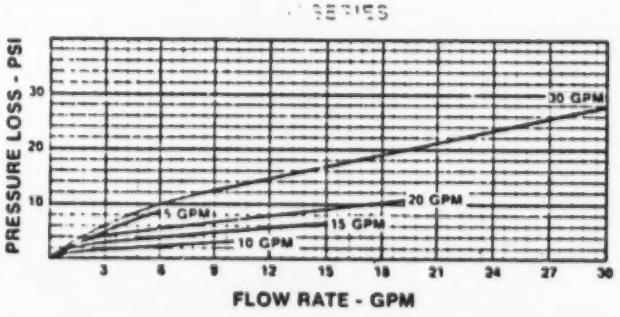
**Scales can be calibrated in other specific gravities for modest additional charge.*

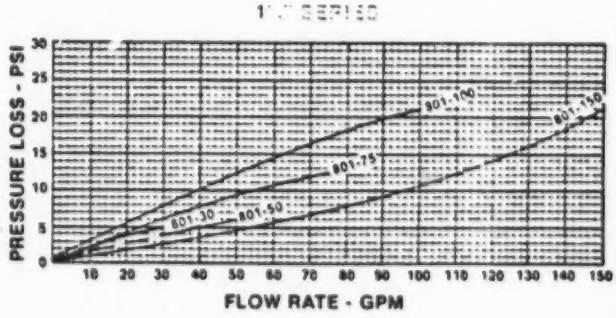
OIL, WATER OR OTHER FLUIDS

APPROXIMATE PRESSURE LOSSES IN FLOW METERS USING OIL AT NORMAL TEMPERATURES
(ADD 10% TO PSI FOR EQUAL WATER FLOWS)

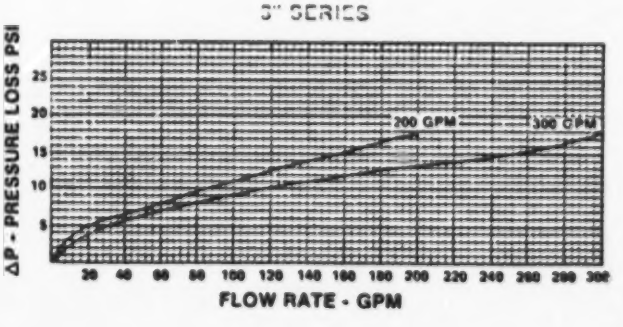


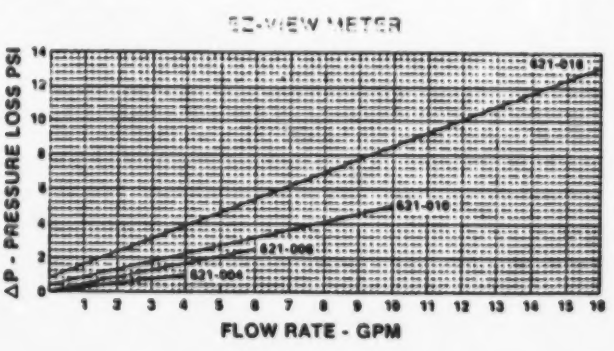






APPROXIMATE PRESSURE LOSSES IN FLOW METERS USING WATER AT NORMAL TEMPERATURES





DEVELOPMENT OF A COMBINED DIESEL EXHAUST GAS COOLER/SCRUBBER FOR USE IN UNDERGROUND OPERATIONS

FINAL REPORT PHASE I

Work on this project is conducted under the auspices of the Canada Centre for Mineral and Energy Technology, Engergy, Mines and Resources, Canada.

DSS File No. 23SQ.23440-2-9180

Canmet Scientific Authority: Mr. E. D. Dainty

October, 1983

MICROMEDIA

TABLE OF CONTENTS

LIST OF TABLES & FIGURES

I Aim

II Summary

III Introduction

IV Conclusions

V Discussion

1. General
2. Calculation Procedures
3. Sample Calculation
4. Exhaust Gas Heat Exchanger Design
5. Scrubber Design Process
6. Cost Analysis
7. Proposed Testing in Phase II

VI References

Appendix "A"

LIST OF TABLES AND FIGURES

Table 1 - Water Consumption Rates for Typical Engines

Table 2 - Comparison of Calculated Tube Side Conductance

Table 3 - Production Cost Estimates

Figure 1 - Graph of Degree of Pre-Cooling vs Water Consumption Rates

Figure 2 - Combined Cooler/Scrubber concept Illustration

I AIM

This project determines the benefits of reducing water consumption by precooling hot engine exhaust when using a water bath type scrubber in gaseous mining environments.

The design calculations for a prototype combined cooler/scrubber for static and operational tests are included.

II SUMMARY

A number of diesel engines in common underground use were studied and their exhaust gas flows and other relevant characteristics compiled.

Water consumption rates and estimated scrubber volumes were derived corresponding to uncooled designs and to designs with precooling of the exhaust gases. The practical limits of exhaust gas cooling were established.

A prototype combined cooler/scrubber has been designed for an engine to be made available by CANMET. The benefits from this approach will be reduced scrubber volume and water consumption.

.../2

III INTRODUCTION

Safe use of diesel-powered mining equipment in underground gassy mines requires that hot engine exhaust be cooled before being released into the mine. Various devices are used to cool gases and remove some particulates in the exhaust. Currently the most common is the water-bath type scrubber. It directs exhaust gases through a water bath with a mist eliminator for entrained water separation and reuse.

In general, these devices all have several disadvantages:

- high water consumption because of the way they operate, since this type of exhaust gas cooling device is based on simple evaporation process.

- The entrained water accumulates in the underground atmosphere and helps to form fog formations which adversely affects visibility.

- There is an increase of exhaust back pressure on the engine which degrades performance.

- There is a possibility of the scrubber running dry.

- The large size of the scrubber affects vehicle design and mobility.

The quantity of water used in a scrubber depends on exhaust gas temperature. By precooling the gases, water consumption can be considerably reduced, eliminating or reducing the disadvantages listed above.

.../3

IV CONCLUSIONS

From this study, it is concluded that:

- The benefits of precooling exhaust gas are in proportion to exhaust gas flow and to temperature.
- The combined cooler/scrubber for the sample engine was smaller than the scrubber required without precooling.
- For the sample engine, with precooling of the exhaust gases from 850°F to 475°F, the water consumption rate dropped from 184 lb/hr to 85 lb/hr.

IV GENERAL DISCUSSION

A study was made of several engines to establish operating parameters and theoretical water consumption rates of an exhaust gas scrubber with and without precooling.

1. Calculation Procedure for Water Consumption Rates

The "Holtz Curve"(1) is based on the theoretical energy required to heat and evaporate water in an exhaust scrubber at sea level. It is used to calculate water consumption based on the volume of exhaust gases, their temperature, and the fuel/air ratio. Graphic presentations accurately represent the practical performance of efficient batch-type conditioners.

Because the "Holtz Curve" is based on dry air, the theoretic volume of water produced by combustion was calculated by balancing the chemical equations of combustion. (See Appendix A for a sample calculation.)

.../4

No correction for barometric pressure was necessary because the engine manufactures supplied information on tests conducted at sea level.

Some of the engines considered for the program are currenty used in underground mines. Information on gas flow, horsepower, etc. was supplied by the manufactures for Caterpillar engines and by EMR (CANMET) for the Ford engine.

The prototype cooled was intended for extended on site testing at the conclusion of this program. Unfortunately, the decision on engine choice was based on availability as no engine in current use could be made available for the test program. However, the Ford engine and all necessary 'hardware' to fit the engine to the dynamometer was made available from CANMET. This engine is a Ford 330 in-line, six cylinder diesel of 100 horsepower. Although this engine is no longer commercially available, it was considered suitable as there would be benefits to precooling of the exhaust gas due to the high gas temperature.

From previous experience with exhaust gas heat exchanger design, it was known that outlet temperatures of 500°F (260°C) could be obtained with a compact unit suitable for mobile equipment. For initial calculations, an outlet temperature that was 60% of inlet temperature was used.

A sample calculation of the procedure used to generate water consumption rates is outlined in the following sections. A more detailed calculation is shown in Appendix "A".

.../5

3. Sample Calculation

Given the following characteristics for the FORD 330 engine:

Inlet Air Rate	= 772	lb/hr
Fuel Flow	= 42.6	lb/hr
.·. F/A ratio	= 0.055	
Temp. Exhaust	= 1010°F	

Because a water jacketed manifold and pipe would be used, the inlet temperature to the scrubber estimated to be 850°F. Assuming a fuel of 13.6%H, ($C_{7.19}$ $H_{13.49}$) the water produced in combustion can be calculated to be 51.8 lb/hr.

The dry gas rate is therefore:

$$M = 772 + 42.6 - 51.8$$
$$= 762.8 \text{ lb/hr.}$$

From the "Holtz Curve" using Tex. = 850°F @0.055 F/A.

T gas out = 148°F
Water consumption = 1.85 U.S. gal/1000 cu. ft.

.·. Dry Gas rate

$$= 762.8/\left((.0752)\left(\frac{460 + 68}{460 + 143}\right) \right)$$

$$= 11{,}920 \text{ cu. ft./hr.}$$

.../6

∴ Water consumption is:

$$1.85 \times \frac{11,920}{1000}$$

= 22.0 US Gal/hr.

or

183.9 lb/hr

Assuming exhaust gas precooling to 475°F; from the "Holtz Curve":

T gas out = 143°F

Water consumption = 0.88 gal/1000 cu. ft.

Dry air flow = $762.8/((.0752)(\frac{460 + 68}{460 + 143}))$

= 11,584 cu. ft./hr.

Water consumption = $\frac{11,584}{1000} \times .88$

= 10.2 U.S. gal./hr.

= 85 lb/hr

Thus pre-cooling reduces water consumption by 54%. Table 1.0 presents engine data and scrubber water consumption rates using the above procedures. Figure 1.0 represents the relationship between water consumption and the degree of cooling. From these examples it is apparent that a 40% reduction in exhaust gas temperature will reduce water consumption by approximately 50%.

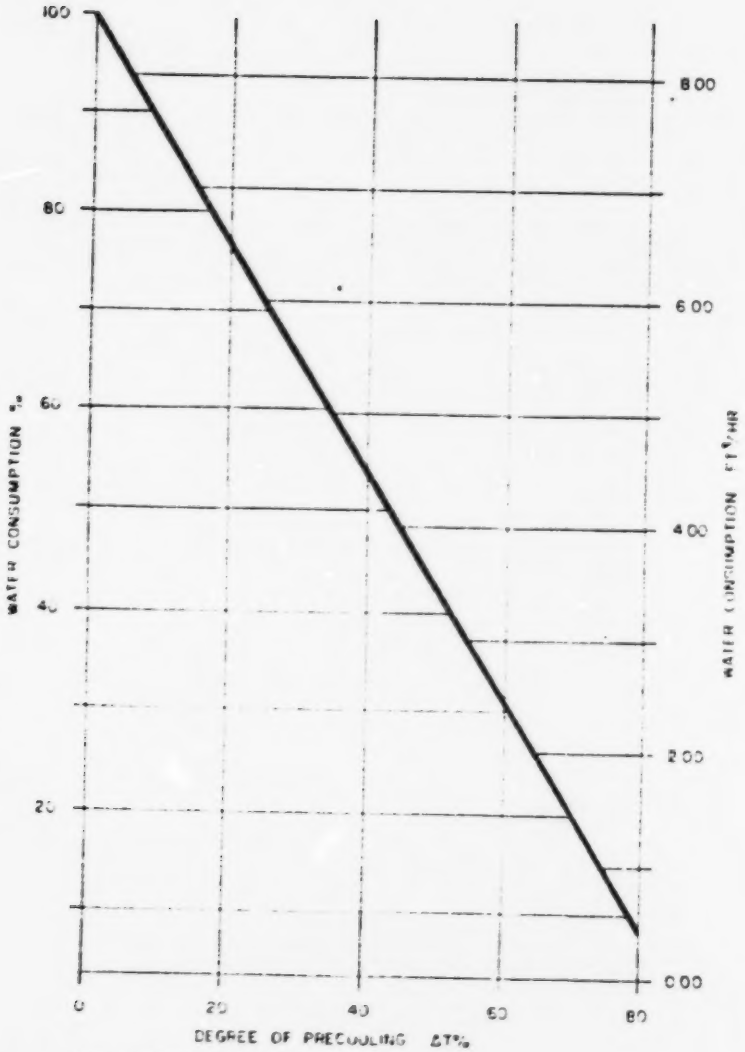
.../7

TABLE I

WATER CONSUMPTION RATES FOR EXAMPLE ENGINES

ENGINE	H.P. @RPM	Airflow #/hr.	Fuel #/hr.	Exhaust Temp °F	Water Cons. #/hr	Exhaust Temp °F Cooled	Water Cons. Precooled #/hr
CATERPILLAR							
3304T	165@ 2200	1843	70	1049	559	630	183
3304 N.A.	100@ 2200	977	43.9	851	225	510	112
3306 N.A.	150@ 2200	1487	65.8	1184	531	710	283
3306T	190@ 2000	2020	77.8	963	512	577	273
3306T	210@ 2000	2309	82.8	968	597	588	319
3306TA	270@ 2200	2886	107.6	1004	818	602	435
FORD 330	97@ 2200	772	42.6	850	184	475	85

.../8



FOR CATERPILLAR 3306NA SEE TABLE 1

FIGURE 1 _ PRECOOLING OF EXHAUST GAS vs REDUCTION OF WATER CONSUMPTION

4. Exhaust Gas Heat Exchanger Design

The basic design configuration of the exhaust gas heat exchanger (cooler) is a single pass, counterflow, tube and shell exchanger with the hot gases on the tube side. This was based on previous design experience at Hovey & Associates (1979) Ltd. and "rules-of-thumb" from various references.

The factors considered included:

a) Fouling - Increased velocities tend to reduce fouling. Straight tubes permit easier mechanical cleaning.

b) Pressure Drop - For the same pressure drop, higher heat transfer coefficients are obtained on the tube-side.

c) Viscosity - Higher heat transfer rates are ordinarily obtained by placing the viscous fluid on the shell side.

d) Tube Size - Under equal velocity conditions, smaller tube diameters increasee the heat transfer coefficient, as well as the pressure drop.

.../10

Design variables include tube size, number, pitch, and length, fluid velocities, and shell side baffling. Mechanical design considerations include tube to tubesheet attachment, ease of maintenance and operating differential expansion between the shell and tubes.

The tube size chosen will provide both the required surface area and a suitably thin wall to allow heat transfer and still permit simple fabrication. Operating temperatures dictate a welded tube to tubesheet attachment. An equilateral triangular pitch was used to provide efficient shell side heat transfer, and the tube pitch was set at the minimum (5/8 inch) as per TEMA Standards.

In order to maintain compact size and effective heat transfer, tube length was limited to 24 inches. This length is compatible with the scrubber design as detailed in Section 3.

Previous experience with an exhaust gas heat exchanger provided some practical limits for gas flow rates required to achieve sufficient heat transfer within reasonable pressure drop limits.

.../11

The gas side flow was set as:

G, gas flux = 21,200 $\frac{\text{lbm}}{\text{hr ft}^2}$.

ΔP , pressure drop, gas side = 5 in. H_2O.

This gas flux sets the number of tubes and hence the heat exchange surface area.

The coolant flow was estimated as 4200 lb/hr. but optimal flow rate will be determined during testing.

The unit conductance on the tube side was calculated using empirical equations (2) (3). The results of several equations are compared in Table 2.

These examples show a fair correlation, although it is not unusual to have a variance of ± 20% between the results obtained by the different equations.

.../12

TABLE 2

COMPARISON OF CALCULATED TUBE SIDE CONDUCTANCE

Source	Equation	Result: h_i, $\frac{BTU}{hr\ ft^2\ °F}$
MCADAMS	$\left(\frac{hD}{k_b}\right) = 0.023 \left(\frac{GD}{\mu_b}\right)^{.08} \left(\frac{\mu c_p}{k}\right)_b^{.04}$	25.93
COLBURN	$\frac{h}{c_{p_b} G} = \left(\frac{\mu_f c_{p_b}}{k_b}\right)^{2/3} = \frac{0.023}{\left(\frac{GD}{\mu_f}\right)^{0.2}}$	26.79
SIEDER-MATE	$\frac{h}{c_{p_b} G} \left(\frac{\mu c_p}{k}\right)_b^{2/3} = \frac{0.027}{\left(\frac{GD}{\mu_b}\right)^{.02}} \left(\frac{\mu_b}{\mu_w}\right)^{.014}$	32.80
SIEDER-MATE	$\bar{h}_i = \frac{k}{D}(0.023)\, Re_d^{.08}\, Pr^{0.33} \left(\frac{\mu_b}{\mu_s}\right)^{0.14}$	27.81* 31.1 (*Corrected)

NOTE: The Above was calculated using temperatures and flow rates approximately equal to those of the sample engine.

.../13

The conductance on the shell side of a tube and shell exchanger is more difficult to calculate as there are few empirical correlations available. Typically, with gas-liquid exchangers the shell side conductance is in the order of 30 times that of the tube side. An inaccuracy in the shell side conductance does not, therefore, appreciably affect the overall conductance.

The design calculation is attached in Appendix "A".

Stress in the tubes and shell due to thermal expansion caused by temperature differences was considered during design. A comparison of the tube side and shell side unit conductances show a ratio of 1:32. The temperature profile through the tube wall will be proportional to that ratio. A tubewall temperature can be estimated:

$$T_{\text{water side}} (200^{\circ}F) + \left(\frac{850^{\circ}F - 200^{\circ}F}{32} \right)$$

$$= 220^{\circ}F$$

A differential temperature of 25°F between the tube wall temperature and the shell will result in negligible longitudinal stress. Fixed tubesheets at both ends of the heat exchanger are therefore suitable.

.../14

The proposed design can be summarized as follows:

Number of Tubes	-	44
Tube size	-	.500 O.D. x .402 I.D. (18 ga. wall)
Tube Pitch	-	Equilateral triangular pitch 5/8 inch centers.
Tube Length	-	24 inches
Coolant-Flow	-	counter-flow
Gas-Flow	-	one pass
Tubesheet attachment	-	welded tubes, shell welded in place

5. Scrubber Design Process

Of the types of water-bath scrubbers currently in use the batch type is more common than the separate water make-up tank type. Previous scrubber design experience at Hovey & Associates (1979) Ltd. has been with batch-type scrubbers. For these reasons, the former type was chosen for this prototype design.

Considerations in the design process include the allowable exhaust gas back pressure, water consumption rate, water depth variation, gas flow passage areas and, in this case, the physical size of the cooler.

The height of a given scrubber is limited by the water consumption rate, by the minimum water depth required, and by the height required for the baffles which remove excess entrained water.

.../15

From these constraints the following can be calculated:

(a) Exhaust backpressure limits:

Maximum allowable exhaust backpressure	=	20 in. H_2O (est.)
Backpressure through cooler	=	5 in. H_2O (est.)
Minimum water depth required	=	5 in. H_2O
∴ allowable water depth variation	=	10 in. H_2O

(b) Physical height limits:

Minimum water depth required	=	5 in.
Water depth variation	=	10 in.
Minimum clearance to first baffle (at maximum water depth)	=	3 in.
Baffle height required	=	8 in.
		26 in.

In the case of this scrubber, the overall height was increased to accommodate the exhaust cooler. This was accomplished by increasing the clearance to the first baffle and the baffle height.

Required cooler length has been calculated to be 24 inches. If the cooler was mounted on top of the scrubber, it would interfere with the scrubber outlet, and would make inlet connections difficult. A vertical mounting position alongside the scrubber would increase the overall height of the combined unit only slightly but would require elaborate jacketed elbow connections at

.../16

the cooler outlet. For these reasons, it was decided to locate the cooler inside the scrubber. Figure 2 illustrates the concept of mounting, gas flow, and water flow for the cooler. However, it is still possible to obtain a combined unit with a compact size, as detailed below.

The scrubber is designed to maintain sufficient water throughout an 8-hour shift operating on a one-third, average load factor. For the Ford engine the scrubber size can be calculated as:

$$\text{Water volume} = 10.2 \text{ U.S. Gal/hr} \times 231 \frac{\text{cu. in.}}{\text{gal}} \times (8 \text{ hrs} \times 1/3 + 10\%) = 7069 \text{ cu. in.}$$

For a depth variation of 10 inches, the volume of the cooler is:

$$= A_0 \times 10$$
$$= 331 \text{ cu. inches.}$$

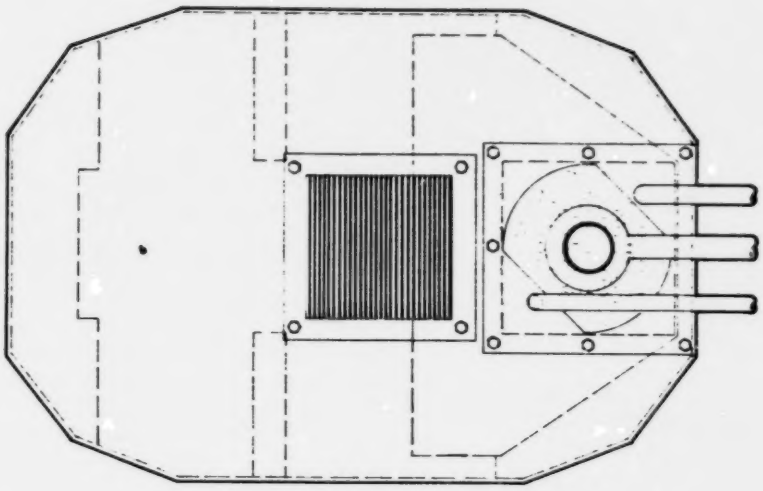
The water space volume in the scrubber is therefore:

$$= 7069 + 331$$
$$= 7400 \text{ cu. in.}$$

The water surface area in the scrubber is therefore:

$$\frac{7400 \text{ cu. in.}}{10 \text{ in. (depth)}} = 740 \text{ sq. in.}$$

.../17



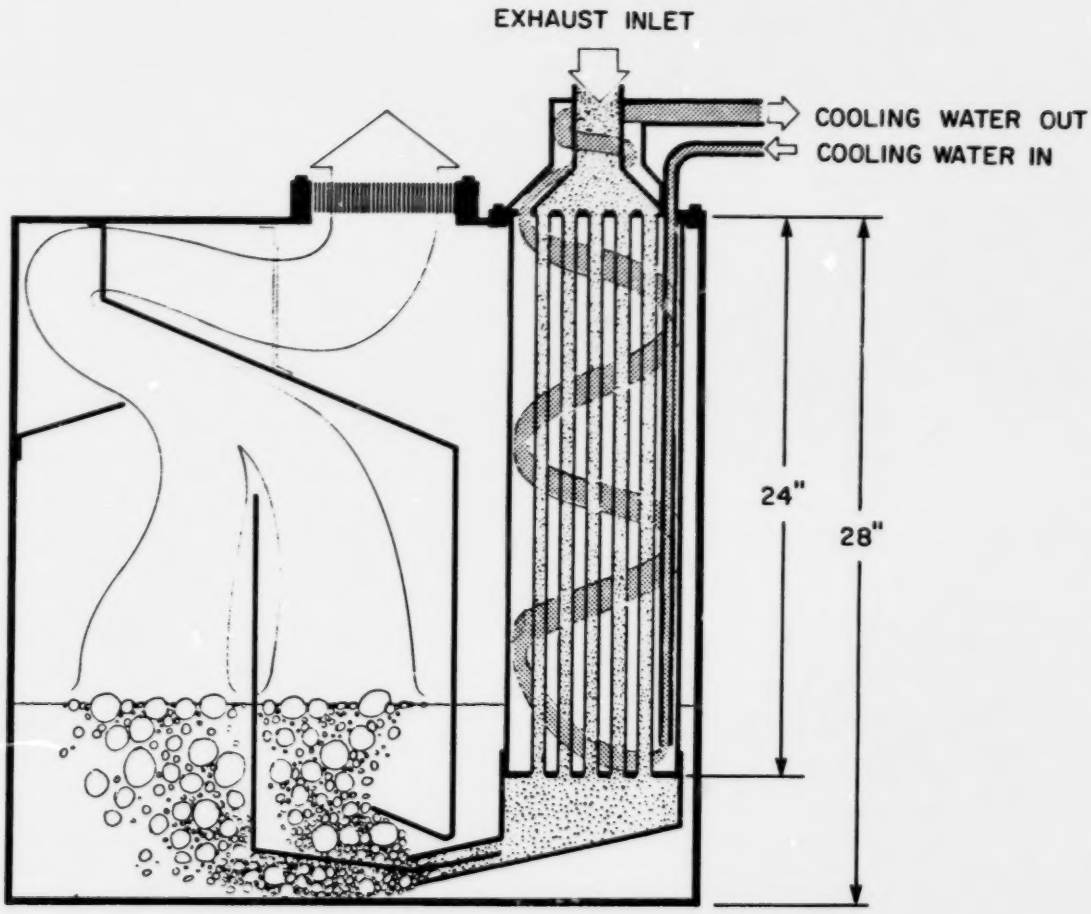


FIGURE 2 COMBINED COOLER SCRUBBER

Based on the above, the physical size of the cooler/scrubber has been defined. The flow passage areas through the various baffles were based on previous experience and calculated by scaling previous designs.

6. Cost Analysis

In order to determine the feasibility of utilizing a cooler/scrubber, cost estimates for construction of various sized units were developed. The costs are based on construction of a production unit (not a prototype) in small quantities (2 to 5 units). Because the sizing techniques for both scrubbers and coolers are relatively well developed, the engineering costs for the first unit will be mainly for drafting services. The cost estimates are presented in Table 3.

From these estimates, it is apparent that the cost of cooler construction will be offset by the smaller scrubber required in most cases. Another potential cost saving, not included in the table, would be the marginally smaller flame trap(s) required.

.../19

TABLE 3

PRODUCTION COST ESTIMATES

Item	Engine Size	Production Cost	Design Cost
Cooler	60 H.P.	$1,500	$1,500
	100 H.P.	2,000	
	150 H.P.	2,300	
Scrubber	60 H.P.	$ 5,000	1,500
	100 H.P.	7,000	
	150 H.P.	10,000	
Combined Cooler/ Scrubber	60 H.P.	$5,500	$2,500
	100 H.P.	7,000	
	150 H.P.	9,300	

.../20

7. PROPOSED TESTING IN PHASE II

The combined cooler scrubber to be constructed in Phase II of the program will be dynamometer tested on the engine at the CANMET facilities. The parameters on the cooler/scrubber to be measured include:

A. Temperature
1. exhaust at engine manifold
2. exhaust into cooler
3. exhaust out of cooler
4. exhaust out of scrubber
5. water (coolant) into cooler
6. water out of cooler
7. water in scrubber
8. wall surface temperature of scrubber

B. Pressure
1. exhaust at engine manifold
2. exhaust into cooler
3. exhaust at cooler outlet
4. exhaust out of scrubber (ambient)
5. ΔP coolant side of cooler

C. Various Water flow rate on coolant side of cooler to optimize water flow.

D. Water consumption rate of scrubber.

A period has been included for modification or rework that may be required. At the conclusion of this testing, an explosive atmosphere certification test is proposed. If the prototype cooler/scrubber can be easily adapted to fit an engine in service, an extended on site test will be included.

REFERENCES

(1) J. C. Holtz; Safety with Mobile Diesel-Powered Equipment Underground

(2) Afgan and Schlunder; Heat Exchangers: Design and Theory Sourcebook

(3) Kakec, Bergles, Mayinger; Heat Exchangers - Thermal - Hydraulic Fundamentals and Design

(4) Waytulonis, Smith, Mejia; Failure Anaysis of Diesel Exhaust-Gas Water Scrubbers

APPENDIX "A"

APPENDIX "A"

Sample Calculation

Dry Gas Rate from Combination Equation.

1. Assumptions: 1. Fuel of 13.6% Hydrogen in the form C_aH_b
   2. No. CO formed (perfect combustion).

$$a = \frac{86.4}{12.01}\ \frac{(\%\ C)}{(\text{Mol. Wt. of C})} = 7.19$$

$$b = \frac{13.6}{1.01} = 13.49$$

Fuel is $C_{7.19}\ H_{13.49}$

Combustion equation is of the form:

$$a(O_2 + 3.76\ N_2) + C_{7.19}H_{13.49} - bO_2 + cN_2 + dCO_2 + eH_2O$$

$$a = b + d + 0.5e$$

$$b = a - d - e/2$$
$$= a - 10.57$$

$$c = 3.76\ a$$

$$e = \frac{13.49}{2} = 6.75$$

$$d = 7.19$$

Equation becomes:

$$a\ (O_2 + 3.76\ N_2) + C_{7.19}H_{13.49} - (a-10.57)O + 2 - 3.76\ aN_2 +$$
$$7.19\ CO_2 + 6.75\ H_2O.$$

total molecular weight of exhaust

$= b + c + d + e$

$= 4.76\ a + 3.37$

mole fraction in exhaust = volume fraction or partial pressure.

mole fraction $H_2O = \frac{6.75}{(4.76\ a + 3.37)}$

$\frac{\text{Mass of Air Flow}}{\text{moles of fuel}} = a\ (32 + 3.76\ (28.02)$

$= 1.373a$

or $a = 0.7283 \times \text{A/F ratio.}$

For this fuel the mol. wt. is:

$(12.01)\ (7.19) + (1.01)(13.49)$
$= 99.98$

2. For the FORD #330 Engine:

Air Flow = 42.6 lb/hr

fuel flow = 772 lb/hr

Exhaust gas temperature = 1010F

It should be noted that the mole fraction of CO_2 in the wet analysis was calculated to be 10.8%, not 11.7% as tested by CANMET.

Number of Moles of Fuel $= \frac{42.6}{99.98} = 0.426$ moles

The water in the gas is therefore

0.426 moles $\times$ 6.75 (mole fraction) $\times$ 18 $\frac{\text{lb}}{\text{mole}}$ = x 51.77 lb water

Exhaust equals:

772 lb air + 42.6 lb. fuel - 51.8 lb water
= 762.8 lb/hr dry gas rate

PART II

APPENDIX "A"

EXHAUST HEAT EXCHANGER CALCULATIONS

FORD 330 Engine

Given: T exhaust = 1010°F, use 850°F inlet to cooler.

m air = 772 lb/hr.

M fuel = 42.6 lb/hr

1.0 Flow Area

Use 44 tubes .500 O.D. x .402 ID.

$A_i = 8.814 \times 10^{-4}$ sq. ft.

$A_T = 0.03878$ sq. ft.

2.0 Seider Tate eq" for tube inner wall conductance.

$$h_i = \frac{k}{D}(0.023)\,Re_d^{0.8}\,Pr^{0.33}\left(\frac{U_b}{U_s}\right)^{0.14}$$

where $Re_d = \frac{GD}{U_b}$

$G = PV$

$= \frac{772 + 42.6}{0.03878} = 21{,}005 \frac{lb}{hr\ ft^2}$

$= 5.835 \frac{lb}{ft^2 sec.}$

$Re_d = \frac{5.835\ (.402)}{(2.03 \times 10^{-5})(12)}$

$= 9629$

.../2

Assume: T bulk = 675°F

U_b, viocosity at 675°F =

U_s, viscoity at 300°F =

k and P_r at 675°F.

$$h_i = \frac{(0.0269)(0.02.023(9629)^{0.8}(0.725)^{0.33}}{.402/12} \frac{(2.03 \times 10^{-5})}{(1.53 \times 10^{-5})}^{0.14}$$

$$= 26.57 \frac{BTU}{hr.\ sq.\ ft.\ ^oF}$$

3. $h_o = 800 \frac{BTU\ (estimated)}{hr.\ sq.\ ft.\ F}$

4. Unit Conductance

$$U_d = 1/\left[\frac{1}{h_o} + Ro + Rk + Ri\ \frac{Ao}{Ai} + \frac{Ao}{Ai}\ \frac{}{hi}\right]$$

$$Rk = \frac{\ln [ro/ri]}{2\pi\ \ 1k}$$

$$= \frac{\ln\ (.500/.402)}{2\pi \left(\frac{24}{12}\right)\ 10.9}$$

$$= 1.59 \times 10^{-3}\ \frac{hr.\ sq.\ ft.\ F}{BTU}$$

$$Ro = 0.001\ \frac{hr.\ sq.\ ft.F}{BTU}\ \text{for water jacket}$$

$$Ri = 0.01\ \frac{hr\ sq.\ ft.F}{BTU}\ \text{for diesel engine exhaust}$$

$$\frac{Ao}{Ai} = \frac{(.500)\ 24}{(.402)\ 24}$$

$$= 1.24$$

$$U_d = 1/\left[\frac{1}{800} + 0.001 + .00159 + .01\ (1.24) + \frac{1.24}{26.57}\right]$$

$$= 15.895\ \frac{BTU}{hr.\ ft.^2F.}$$

5. Calculate exhaust gas temp at outlet

$Ao = 44 \times (.500) \times 24$

$= 11.52\ ft^2$

$UAo = 183.1\ \frac{BTU}{hrF}$

$$LMTD = \frac{Ta - Tb}{\ln (Ta/Tb)}$$

$$= \frac{(850-180) - (500-200)}{\ln (670/300)}$$

$= 460^oF.$

$q = (UAo)\ LMTD$

$= (183.1)(460)$

$= 84,269$ BTU/hr.

for spec. ht. of exhaust gas at 670°F,

$C_p = 0.266$

$M\ air\ (C_p)(\Delta T) = q$

$(814.6)(.266)\ T = 84,269$

$\Delta T = 389^oF.$

$T\ out = 850-389 = 461^oF.$

Iterating

$$LMTD = \frac{(850 - 180) - (475 - 200)}{\ln(670/275)}$$

$= 444^oF$

$q = 183.1 \times 444^oF$

$= 81,217\ \frac{BTU}{hr.}$

$M\ air\ C_p\ T = 81,217$

$(814.6)\ (.266)\ T = 81,217$

$T = 375^oF$

$850 - 375^o = 475^oF$ outlet temp.